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COMPONENT PERFORMANCE INVESTIGATION OF J71 EXPERIMENTAL TURBINE

IV - EFFECT OF FIRST-STATOR ADJUSTMENT; OVER-ALL PERFORMANCE OF J71-97 TURBINE WITH 132-PERCENT-DESIGN STATOR AREA

By Elmer H. Davison, Donald A. Petrash, and Harold J. Schum

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COMPONENT PERFORMANCE INVESTIGATION OF J71 EXPERIMENTAL TURBINE

IV - EFFECT OF FIRST-STATOR ADJUSTMENT; OVER-ALL PERFORMANCE OF J71-97 TURBINE WITH 132-PERCENT-DESIGN STATOR AREA

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SUMMARY

The performance changes resulting from adjusting the first-stator area of the J71 experimental three-stage turbine are being investigated. The performance of this turbine with stator areas 97 percent of design has been presented previously. The performance of this same turbine with the first-stator area increased to 132 percent of design by changing the stagger angle of the blades is presented herein.

Comparison of the performance of these two turbines showed that (1) the maximum efficiency obtained decreased from 0.891 for the J71-97 turbine to 0.869 for the J71-132 turbine; (2) the initial choke point changed from the third-stage rotor for the 97-percent turbine to the first-stage rotor of the 132-percent turbine, with subsequent blade rows choking as the over-all pressure ratio was increased; and (3) increasing the first-stator throat area from 97 to 132 percent of design increased the value of the choking equivalent weight flows by approximately 13.6 percent. In addition to the performance comparisons, example turbine match points based on the experimental data were calculated for an engine mode of operation during which the compressor is maintained at constant design equivalent conditions. From these calculations it was concluded that (1) increasing the first-stator area from 97 to 132 percent of design corresponds to increasing the engine temperature ratio (turbine inlet to compressor inlet) 27.5 percent; and (2) the turbine efficiency at the match point decreased from 0.87 to 0.86 for this increase in the first-stator area.

INTRODUCTION

The NACA Lewis laboratory is currently conducting a study of the performance changes resulting from adjusting the first-stator area of a high-work-output low-speed multistage turbine. If a turbine is operated with fixed first-stator area, the choking inlet equivalent weight flow of the turbine is either constant or has only a small variation. This restriction of the inlet equivalent weight flow limits the range of engine temperature ratio (turbine inlet to compressor inlet) over which an engine can operate. Several of the advantages obtained from increasing the engine temperature-ratio range by stator adjustment are (1) decreased acceleration time and (2) added flexibility in the mode of engine operation.

An analytical investigation was made in reference 1 of the performance changes resulting from stator-area adjustment for a particular three- 3837
stage turbine. This investigation showed that increases in equivalent weight flow were smaller than corresponding increases in stator throat area for adjustment of the first-stator area or simultaneous adjustment of all three stator areas. In addition, the percent increase in inlet equivalent weight flow obtained by adjusting all three stators simultaneously was nearly twice that obtained by adjusting only the first stator.

The purpose of this report is to evaluate the effect on over-all performance of increasing the first-stator throat area of the J71-97 experimental three-stage turbine (ref. 2) to 132 percent of its design value. This modified turbine, hereinafter called the J71-132 or simply the 132-percent turbine, was investigated as a component under equivalent cold-air operating conditions. The increase in stator throat area was obtained by changing the stagger angle of the design blade profiles.

The new compressor and turbine match points determined by a given stator area change depend on the mode of engine operation selected (operation with constant rotative speed, operation with constant exhaust-nozzle area, maintaining the compressor at equivalent design conditions, etc.). The best mode of engine operation for a particular engine over a range of flight conditions can only be determined from an extensive cycle analysis. No attempt is made in this report to evaluate the advantages of the different modes. However, a turbine match point for the 132-percent stator setting was calculated based on the assumption that the compressor continued to operate at design equivalent conditions. Thus, the change in turbine performance was evaluated for at least one mode of engine operation.

SYMBOLS

The following symbols are used in this report:

b ratio of bleed air to compressor weight flow

E enthalpy drop based on torque measurements, Btu/lb

f fuel-air ratio

g gravitational constant, 32.174 ft/sec^2

J mechanical equivalent of heat, 778 ft-lb/Btu

N rotational speed, rpm

p pressure, in. Hg abs

p_x' rating total pressure, static pressure plus velocity pressure corresponding to axial component of velocity, in. Hg abs

R gas constant, 53.4 ft-lb/(lb)($^{\circ}$R)

T temperature, $^{\circ}$R

w weight flow, lb/sec

$\frac{wN}{60\delta}\,\epsilon$ weight-flow parameter based on product of equivalent weight flow and equivalent rotor speed

γ ratio of specific heats

δ ratio of inlet-air pressure to NACA standard sea-level pressure, $p_0'/29.92$ in. Hg abs

ϵ function of γ, $\frac{\gamma_{sl}}{\gamma_e}\left[\frac{\left(\frac{\gamma_e+1}{2}\right)^{\frac{\gamma_e}{\gamma_e-1}}}{\left(\frac{\gamma_{sl}+1}{2}\right)^{\frac{\gamma_{sl}}{\gamma_{sl}-1}}}\right]$

η_i brake internal efficiency, ratio of actual turbine work based on torque measurements to ideal turbine work based on inlet total pressure p_0' and outlet rating total pressure $p_{x,7}'$

θ_{cr} squared ratio of critical velocity at NACA standard sea-level temperature of 518.7° R, $\dfrac{\frac{2\gamma}{\gamma+1}gRT_0'}{\frac{2\gamma_{sl}}{\gamma_{sl}+1}gRT_{sl}}$

τ torque, ft-lb

Subscripts:

C	compressor
d	design
e	engine operating conditions
sl	NACA standard sea-level conditions
T	turbine 3837
x	axial
0,1,2, 3,4,5, 6,7	measuring stations (see fig. 2)

Superscript:

'	total or stagnation state

APPARATUS AND PROCEDURE

Both this investigation and that of the J71-97 turbine (ref. 2) were carried out with the same turbine test installation. A photograph of the over-all turbine experimental setup is shown in figure 1.

For this investigation the first stator of the J71-97 turbine was replaced with one having design blade profiles set at the stagger angle required to increase the throat area to 132 percent of the design area. The area was limited to a maximum of 132 percent of design because any larger area moves the throat upstream of the trailing edge at the mean blade section.

The instrumentation used in the investigation was the same as that described in reference 2. A schematic diagram of the turbine showing the instrumentation is presented in figure 2. Measurements of total pressure, wall static pressure, and total temperature were taken at the turbine inlet (station 0) and at the turbine outlet (station 7). In addition, wall static taps were installed on both the inner and outer shrouds ahead of each row of blades and at the turbine outlet.

For the turbine with design stator areas, the equivalent design conditions are as follows:

Work, Btu/lb 32.4
Weight flow, lb/sec 40.3
Rotational speed, rpm 3028
Engine temperature ratio (turbine inlet to compressor inlet) 4.16

The turbine was operated at a nominal inlet pressure p_0' and temperature T_0' corresponding to 33 inches of mercury absolute and 700° R. A range of rating total-pressure ratio $p_0'/p_{x,7}'$ from 1.4 to 4.2 was imposed across the turbine, and the speed was varied from 20 to 130 percent of equivalent design speed $N/\sqrt{\theta_{cr}}$.

The values of equivalent weight flow have been corrected for the fuel addition required to maintain the 700° R turbine-inlet temperature. Turbine efficiency is based on measured torque, weight flow, and the turbine-outlet pressure $p_{x,7}'$, which was calculated by adding the axial component of the velocity pressure to the average wall static pressure at the turbine-discharge measuring station. The axial component of velocity was computed from the turbine weight flow (air flow plus fuel flow), the known annular area at the measuring station, and the measured total pressure, total temperature, and total- to static-pressure ratio. The calculated outlet pressure charges the turbine for the energy of the rotor-discharge tangential velocity.

A turbine match point was calculated based on the assumption that the compressor continued to operate at design equivalent conditions. The method used to determine this match point is given in the appendix.

RESULTS AND DISCUSSION

Over-All Performance

The over-all performance of the 132-percent turbine is presented in figure 3 as a plot of equivalent work E/θ_{cr} against the flow parameter $(wN/60\delta)\epsilon$ for constant values of equivalent speed $N/\sqrt{\theta_{cr}}$ and rating total-pressure ratio $p_0'/p_{x,7}'$. In addition, contours of constant brake internal efficiency η_i based on torque and rating pressure ratio are shown.

In general, the performance was good. The maximum efficiency obtained was 0.869, occurring at 120 percent of equivalent design speed and an equivalent shaft work of 32.4 Btu per pound. This compares favorably with the maximum efficiency of 0.891 obtained with the turbine of reference 2, which had stator areas approximately 97 percent of design.

The variation of equivalent torque with rating total-pressure ratio for the equivalent speeds investigated is shown in figure 4. Limiting loading was not obtained with the pressure ratios imposed across the turbine, as evidenced by the continual increase of torque with pressure ratio 3837
at all speeds investigated. The same characteristics were observed for the turbine of reference 2.

Choking Characteristics

The variation of equivalent weight flow with rating total-pressure ratio for the equivalent speeds investigated is shown in figure 5. Choking weight flow, indicated when the curves have a zero slope, was obtained for speeds of 70 percent of design equivalent speed and above. The value of choking weight flow decreased with an increase in speed, which indicates that the first stator is not choked but that some blade row downstream of this stator is choked.

The increase in the value of the choking weight flow obtained by increasing the first-stator area from 97 (ref. 2) to 132 percent of design is shown in figure 6. The choking equivalent weight flow (maximum weight flow) as a percent of the equivalent design weight flow is plotted against percent equivalent design speed for those speeds at which choking was obtained. At design equivalent speed, figure 6 shows that this increase in the first-stator area resulted in a 13.6-percent increase in the choking weight flow. Approximately the same percentage increase in the choking weight flow occurred at the other speeds.

Increasing the throat area of the first stator corresponds to operating the engine at a higher engine temperature ratio (turbine inlet to compressor inlet). For nearly any mode of engine operation specified, this would cause the turbine equivalent speed to decrease or perhaps to remain constant. It is also doubtful that the equivalent speed of the turbine would drop much below 80 percent of the equivalent design speed. For these reasons, the choking characteristics of the turbine at 80 and 100 percent of equivalent design speed have been examined in detail and compared with the choking characteristics for the turbine of reference 2.

The static-pressure distribution at the hub of the blades is plotted against rating total-pressure ratio $p_0'/p_{x,7}'$ for 80 and 100 percent of equivalent design speed in figures 7(a) and (b), respectively. The static pressure at each station has been divided by the inlet total pressure in order to eliminate the effect of the small fluctuations in inlet total pressure encountered while testing the turbine. These figures are useful in determining the choking characteristics of the turbine.

Choking in a blade row, or downstream of the blade row, is indicated when the ratio of the static to inlet total pressure p/p_0' ahead of the blade row remains constant with increasing rating pressure ratio $p_0'/p_{x,7}'$.

Choking in a given blade row rather than some point downstream of the blade row occurs if the pressure-ratio curve ahead of the blade row levels out at a lower rating pressure ratio than those curves at measuring stations farther downstream. Based on this criterion, figure 7(a) for 80 percent of equivalent design speed indicates that the first-stage rotor, second-stage stator, third-stage stator, and third-stage rotor choke successively as the rating pressure ratio increases, since the pressure-ratio curves at stations 2, 3, 5, and 6 level off at increasingly higher rating pressure ratios. The pressure-ratio curve at station 4, ahead of the second-stage rotor, appears to level out at about the same rating pressure ratio as at station 5. Therefore, it cannot be definitely established from figure 7(a) whether the second-stage rotor has choked, although it may choke simultaneously with the third-stage stator. At the entrance to the first stator, station 1, the Mach numbers are quite low over the entire range of rating pressure ratio. Thus, the static-pressure variation at this station is quite small, which makes it difficult to establish from figure 7(a) whether the first stator chokes. However, the previous examination of the weight-flow curves of figure 5 showed that this stator did not choke. The choking pattern of the turbine at 100 percent of design equivalent speed (fig. 7(b)) appears to be the same as at 80 percent of design equivalent speed.

For the turbine with 97 percent of design area stator (ref. 2), it was only established that the last rotor choked. Other blade rows, with exception of the first stator, may have choked simultaneously with the last rotor, but this could not be established. Because the equivalent weight flow varied with equivalent speed, it was known that the first stator did not choke. Thus, opening up the first stator of the turbine moved the initial choke point into the first-stage rotor with subsequent blade rows choking as the over-all pressure ratio was increased.

Match Point

A turbine match point based on the experimental data was calculated for an engine mode of operation during which the compressor is maintained at constant design equivalent conditions. Other modes of engine operation would result in different turbine match points that perhaps would result in better over-all engine performance for the same thrust output.

However, the match point calculated will serve to show the typical change in turbine performance that can be expected when the throat area of the first stator is increased to 132 percent of design by changing the stagger angle of the blades. The design conditions, along with the experimental match point for the 97-percent-area turbine (ref. 2) and the 132-percent-area turbine of this report, are listed in the following table:

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	Design	Turbine of ref. 2, 97-percent-design stator areas	Turbine of ref. 2 with first-stator area increased to 132-percent design
Equivalent work, Btu/lb	32.4	29.7	23.2
Equivalent weight flow, lb/sec	40.3	42.0	47.3
Equivalent speed, percent design	100	96	85.2
Experimental turbine efficiency		0.87	0.86
Engine temperature ratio (turbine inlet to compressor inlet)	4.16	4.51	5.75

These data were obtained from figures 3 and 8 of this report and the experimental performance map, figure 4 of reference 2. The data in this table show that the increase in the first-stator area from 97 to 132 percent of design resulted in a 27.5-percent increase in engine temperature ratio.

As noted previously, the maximum efficiency of the 132-percent turbine was 0.869, which compared favorably with the maximum of 0.891 for the 97-percent turbine (ref. 2). At the match points, the efficiency decreased from 0.87 for the 97-percent turbine to 0.86 for the 132-percent turbine. The match point for the 132-percent turbine, therefore, remained in the region of good efficiency. At the match points, the equivalent weight flow of the 132-percent turbine was approximately 12.6 percent higher than for the 97-percent turbine.

SUMMARY OF RESULTS

From an investigation of the J71-97 experimental three-stage turbine with the first-stator area increased to 132 percent of design by changing the stagger angle of the blades, the following results were obtained:

1. The maximum efficiency obtained was 0.869, occurring at 120 percent of equivalent design speed and an equivalent shaft work of 32.4 Btu per pound. This compared favorably with the maximum efficiency of 0.891 obtained with the turbine investigated previously which had stator areas approximately 97 percent of design.

2. For the turbine with 97-percent-design-area stators, it appeared that only the third-stage rotor choked. Opening the first stator of the turbine moved the initial choke point into the first-stage rotor, with subsequent blade rows choking as the over-all pressure ratio was increased.

3. Increasing the first-stator throat area from 97 to 132 percent of design resulted in approximately a 13.6-percent increase in the choking equivalent weight flows of the turbine.

4. Limiting blade loading was not reached over the range of conditions investigated for either turbine.

5. Example turbine match points based on the experimental data were calculated for an engine mode of operation during which the compressor is maintained at constant design equivalent conditions. From these calculations, the following conclusions were obtained:

(a) Increasing the first-stator area from 97 to 132 percent of design corresponded to increasing the engine temperature ratio (turbine inlet to compressor inlet) by 27.5 percent.

(b) The turbine efficiency at the match points decreased from 0.87 to 0.86 for an increase in the first-stator area from 97 to 132 percent of design.

(c) The turbine equivalent weight flow at the match points increased approximately 12.6 percent for an increase in the first-stator area from 97 to 132 percent of design.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, August 9, 1955

APPENDIX - MATCH-POINT SELECTION

The turbine match point was calculated for an engine mode of operation during which the compressor is maintained at constant equivalent design conditions. In the following discussion, the subscripts 1, 2, and 3 refer to the compressor inlet, compressor outlet, and turbine inlet, respectively. The compressor parameters $E_C/\theta_{cr,1}$, $N_C/\sqrt{\theta_{cr,1}}$, $\frac{w_C\sqrt{\theta_{cr,1}}}{\delta_1}\epsilon_1$, and p_2'/p_1' were assumed to be constant, because the compressor is maintained at constant design equivalent conditions. The pressure ratio across the burners p_3'/p_2' was also assumed constant. 3837

The three matching relations between the compressor and turbine in the engine can be written as follows:

Rotative speed:

$$N_T = N_C \tag{1}$$

Continuity:

$$w_T = (1 + f)(1 - b)\ w_C \tag{2}$$

Work:

$$E_T = \frac{1}{(1 + f)(1 - b)} E_C \tag{3}$$

where equation (1) results because the compressor and turbine are directly coupled.

In the following development it is assumed that the fuel-air ratio f and the bleed-air ratio b are such that $(1 + f)(1 - b)$ always has a value of unity. Equations (1) to (3) can be written in terms of equivalent component operating conditions as follows:

$$\frac{N_C}{\sqrt{\theta_{cr,1}}} = \frac{N_T}{\sqrt{\theta_{cr,3}}}\sqrt{\frac{\theta_{cr,3}}{\theta_{cr,1}}} \tag{4}$$

$$\left(\frac{w_C\sqrt{\theta_{cr,1}}}{\delta_1}\right)\epsilon_1 = \left(\frac{w_T\sqrt{\theta_{cr,3}}}{\delta_3}\right)\epsilon_3\sqrt{\frac{\theta_{cr,1}}{\theta_{cr,3}}}\,\frac{p_3'}{p_2'}\,\frac{p_2'}{p_1'}\,\frac{\epsilon_1}{\epsilon_3} \tag{5}$$

$$\frac{E_C}{\theta_{cr,1}} = \frac{E_T}{\theta_{cr,3}}\,\frac{\theta_{cr,3}}{\theta_{cr,1}} \tag{6}$$

If the ratios of specific heats γ_1 and γ_3 as well as $E_C/\theta_{cr,1}$, $N_C/\sqrt{\theta_{cr,1}}$, $(w_C\sqrt{\theta_{cr,1}}/\delta_1)\ \epsilon_1$, p_2'/p_1', and p_3'/p_2' are assumed constant, it follows from equations (4) to (6) that

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$$\frac{N_T}{\sqrt{\theta_{cr,3}}} = \left(\frac{N_T}{\sqrt{\theta_{cr,3}}}\right)_d \left(\sqrt{\frac{T_3'}{T_1'}}\right)_d \sqrt{\frac{T_1'}{T_3'}} \tag{7}$$

$$\frac{w_T\sqrt{\theta_{cr,3}}}{\delta_3}\ \epsilon_3 = \left(\frac{w_T\sqrt{\theta_{cr,3}}}{\delta_3}\ \epsilon_3\right)_d \left(\sqrt{\frac{T_1'}{T_3'}}\right)_d \sqrt{\frac{T_3'}{T_1'}} \tag{8}$$

$$\frac{E_T}{\theta_{cr,3}} = \left(\frac{E_T}{\theta_{cr,3}}\right)_d \left(\frac{T_3'}{T_1'}\right)_d \frac{T_1'}{T_3'} \tag{9}$$

The turbine design values of $N_T/\sqrt{\theta_{cr,3}}$, $(w_T\sqrt{\theta_{cr,3}}/\delta_3)\ \epsilon_3$, $E_T/\theta_{cr,3}$, and the design engine temperature ratio T_3'/T_1' are given in the APPARATUS AND PROCEDURE section of this report. With the design values known, the turbine parameters $(w_T\sqrt{\theta_{cr,3}}/\delta_3)\ \epsilon_3$, $E_T/\theta_{cr,3}$, $(w_T N_T/60\delta_3)\ \epsilon_3$, and engine temperature ratio T_3'/T_1' can all be plotted against percent design equivalent speed $\dfrac{N_T}{\sqrt{\theta_{cr,3}}}\Big/\left(\dfrac{N_T}{\sqrt{\theta_{cr,3}}}\right)_d$ as shown in figure 8. Any set of turbine parameter values satisfying these conditions will, for the engine temperature ratio indicated, maintain the compressor at design equivalent conditions.

The experimental match point for the 132-percent turbine that satisfies these conditions was determined in the following manner. With the compressor operating at constant design equivalent conditions, it can be shown that the equivalent torque of the turbine $(\tau_T/\delta_3)\ \epsilon_3$ is constant at the design value. The turbine equivalent torque can be written as follows:

$$\frac{\tau_T}{\delta_3}\ \epsilon_3 = \frac{E_T}{\theta_{cr,3}}\ \frac{w_T\sqrt{\theta_{cr,3}}}{\delta_3}\ \epsilon_3\ \frac{\sqrt{\theta_{cr,3}}}{N_T}\ \frac{60J}{2\pi} \tag{10}$$

or, upon substitution of equations (4) to (6), as

$$\frac{\tau_T}{\delta_3}\ \epsilon_3 = \frac{E_C}{\theta_{cr,1}}\ \frac{w_C\sqrt{\theta_{cr,1}}}{\delta_1}\ \epsilon_1\ \frac{\sqrt{\theta_{cr,1}}}{N_C}\ \frac{p_2'}{p_3'}\ \frac{p_1'}{p_2'}\ \frac{60J}{2\pi}\ \frac{\epsilon_3}{\epsilon_1} \tag{11}$$

Since $E_C/\theta_{cr,1}$, $(w_C\sqrt{\theta_{cr,1}}/\delta_1)\,\epsilon_1$, $N_C/\sqrt{\theta_{cr,1}}$, p_3'/p_2', p_2'/p_1', γ_1, and γ_3 were assumed constant, the equivalent torque of the turbine is constant at the design value. At design torque, the over-all rating total-pressure ratio at the different percentages of design equivalent speeds can be obtained from figure 4. The equivalent weight flow at these pressure ratios and speeds can then be determined from figure 5. A plot of equivalent weight flow against percent design equivalent speed for constant design equivalent torque can thus be obtained. Such a plot from the experimental data of the 132-percent turbine is shown superimposed on the first plot of figure 8. The intersection of this curve with the solid curve determines 3837
the match point of the turbine for constant compressor design equivalent conditions. This match point is also indicated on the turbine performance map (fig. 3). The design match point and the match point for the experimental test data of reference 2 are also shown in figure 8.

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1. Rohlik, Harold E., and Rebeske, John J.: Acceleration of High-Pressure-Ratio Single-Spool Turbojet Engine As Determined from Component Performance Characteristics. III - Effect of Turbine Stator Adjustment. NACA RM E54F04, 1954.

2. Petrash, Donald A., Schum, Harold J., and Davison, Elmer H.: Component Performance Investigation of J71 Experimental Turbine. III - Effect of Third-Stage Shrouding on Over-All Performance. NACA RM E55C29, 1956.

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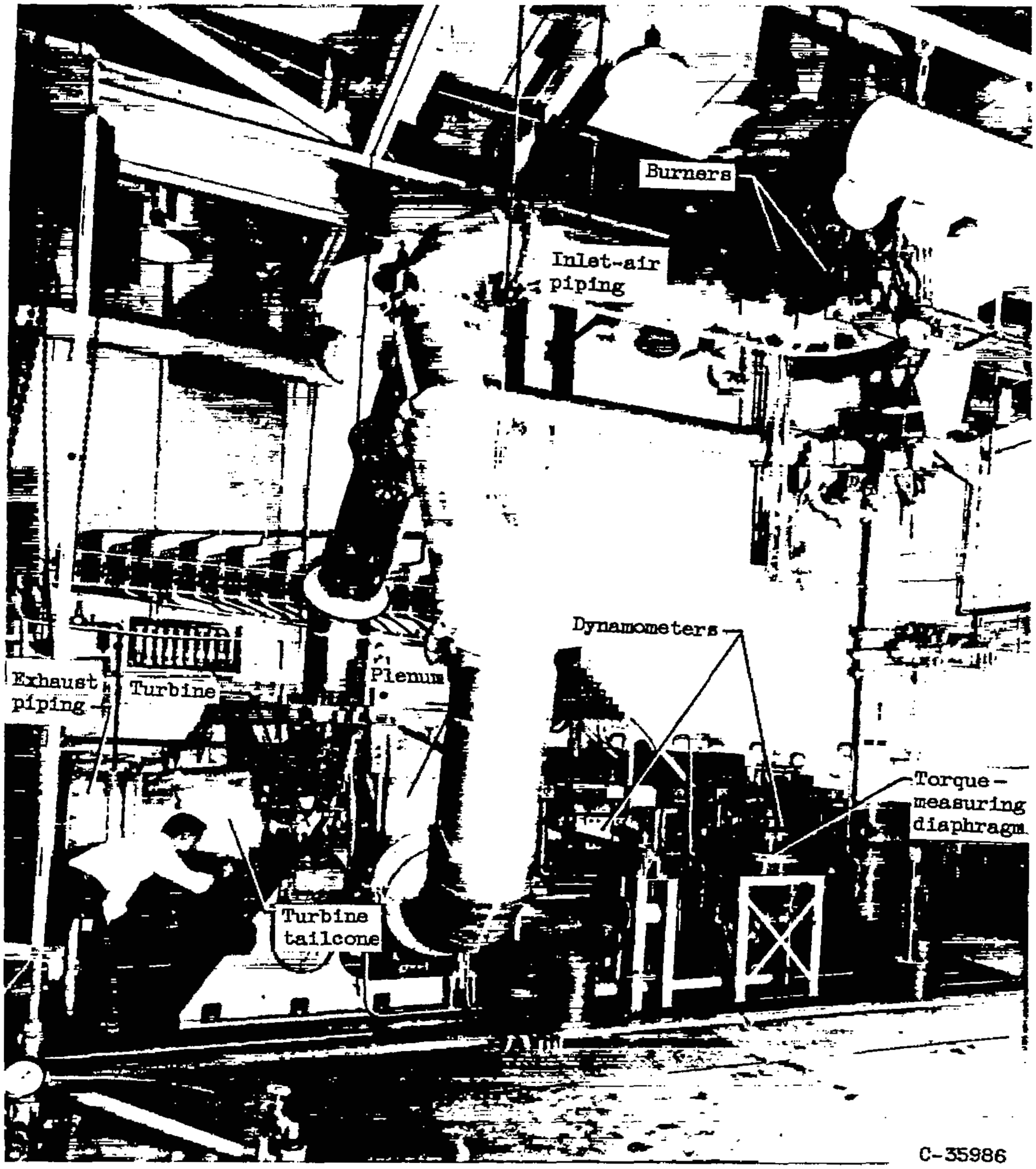

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Figure 1. - Installation of J71-132 experimental three-stage turbine in full-scale turbine component test facility.

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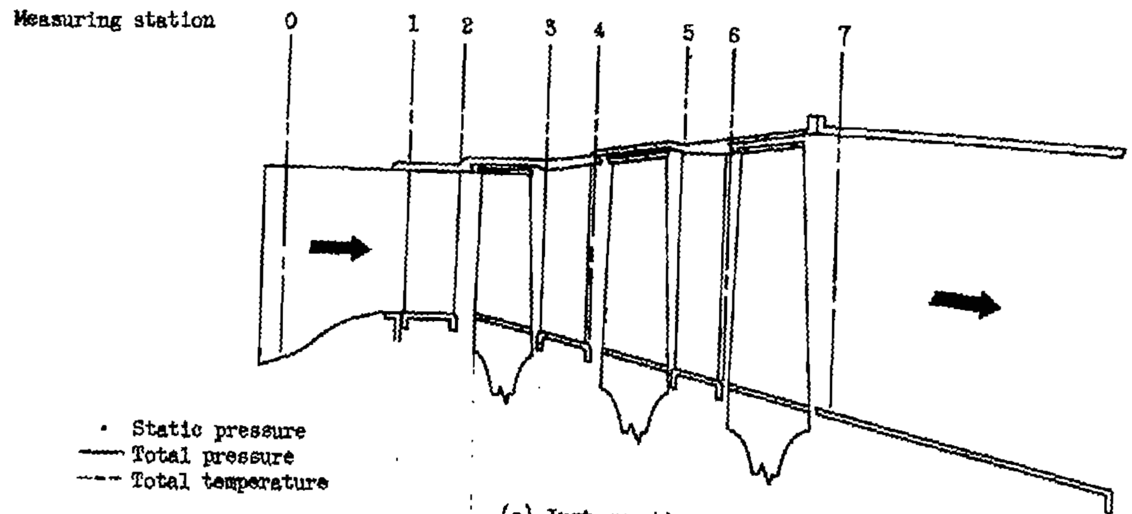
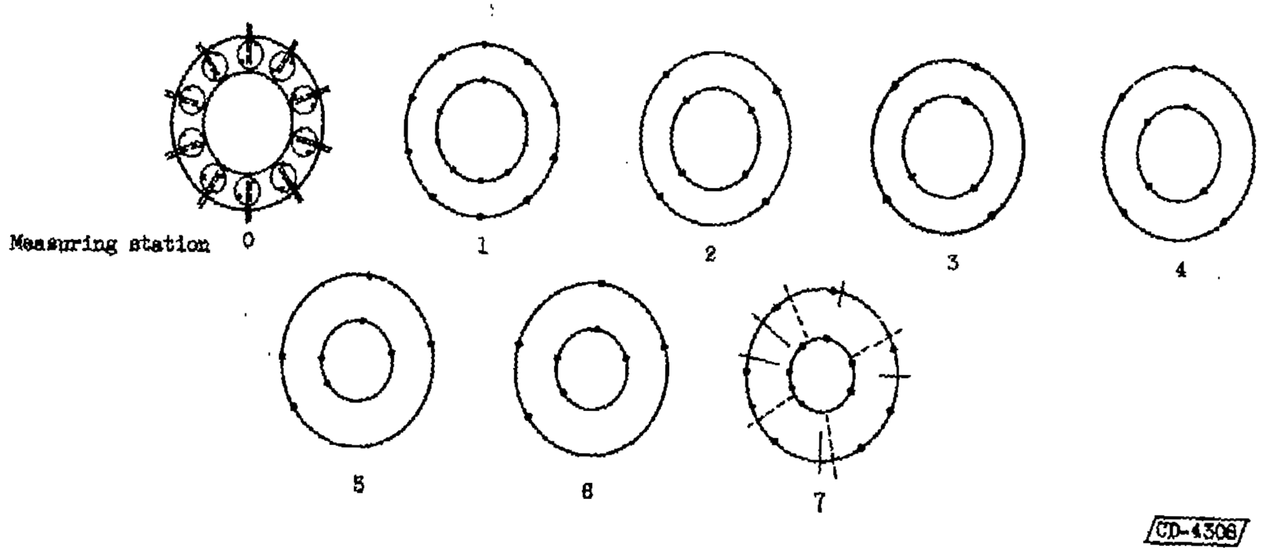


(a) Instrumentions stations.



(b) Circumferential location of instruments at each station.

Figure 2. - Schematic diagram of J71-132 experimental turbine showing instrumentation.

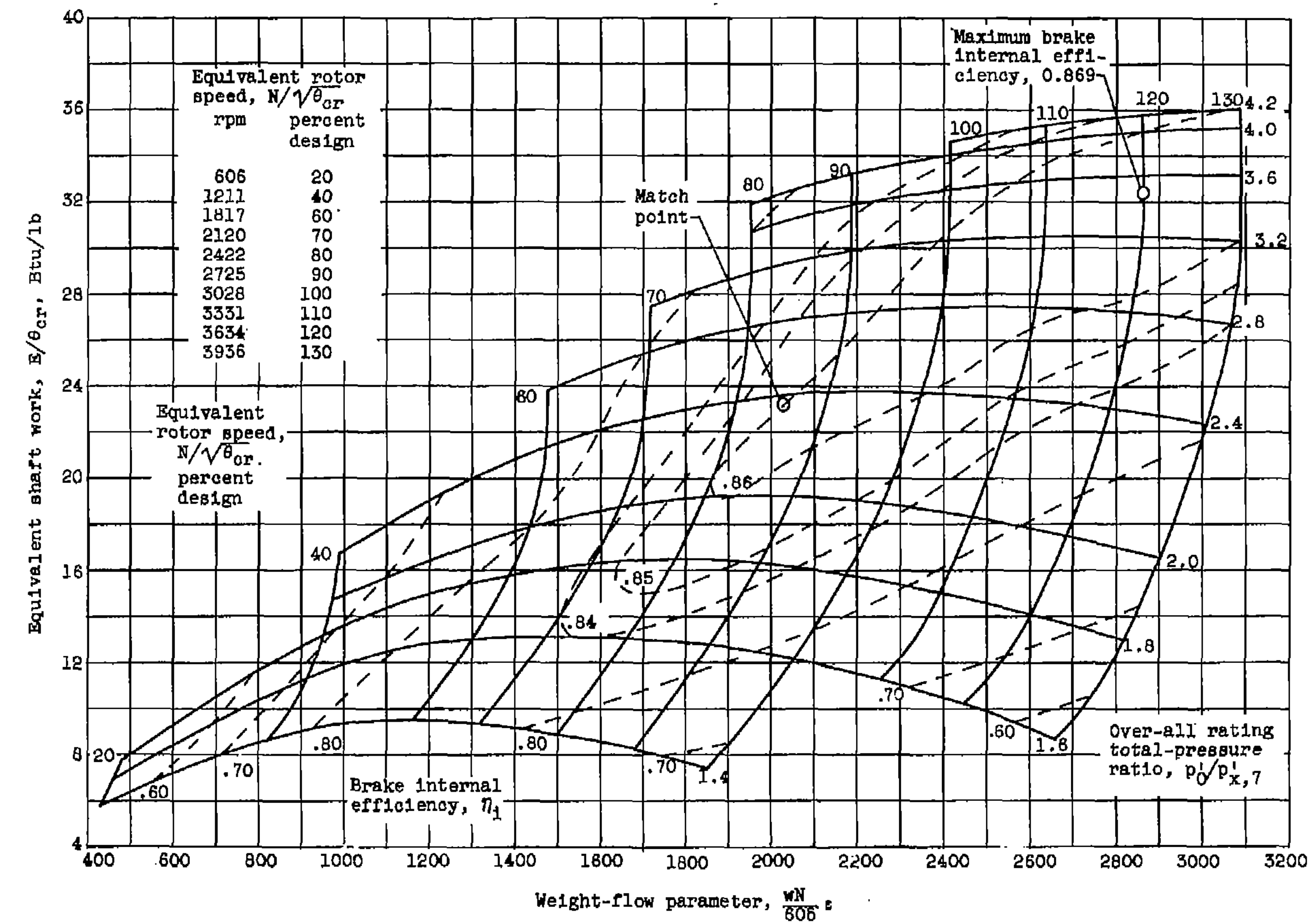


Figure 3. - Over-all performance of J71-132 experimental turbine. Turbine-inlet pressure, 33 inches of mercury absolute; turbine-inlet temperature, 700° R.

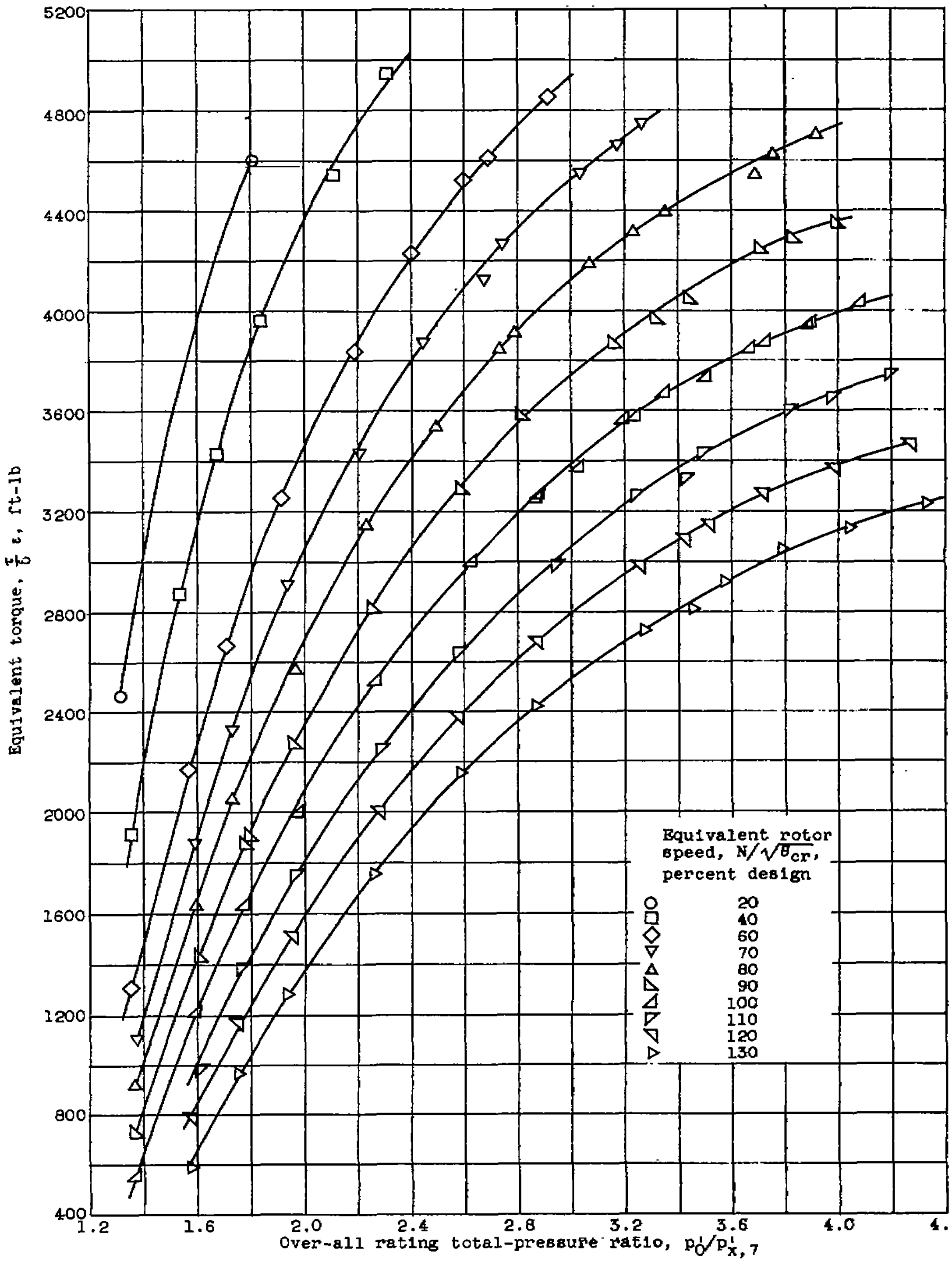


Figure 4. - Variation of equivalent torque with over-all rating total-pressure ratio for values of constant equivalent rotor speed.

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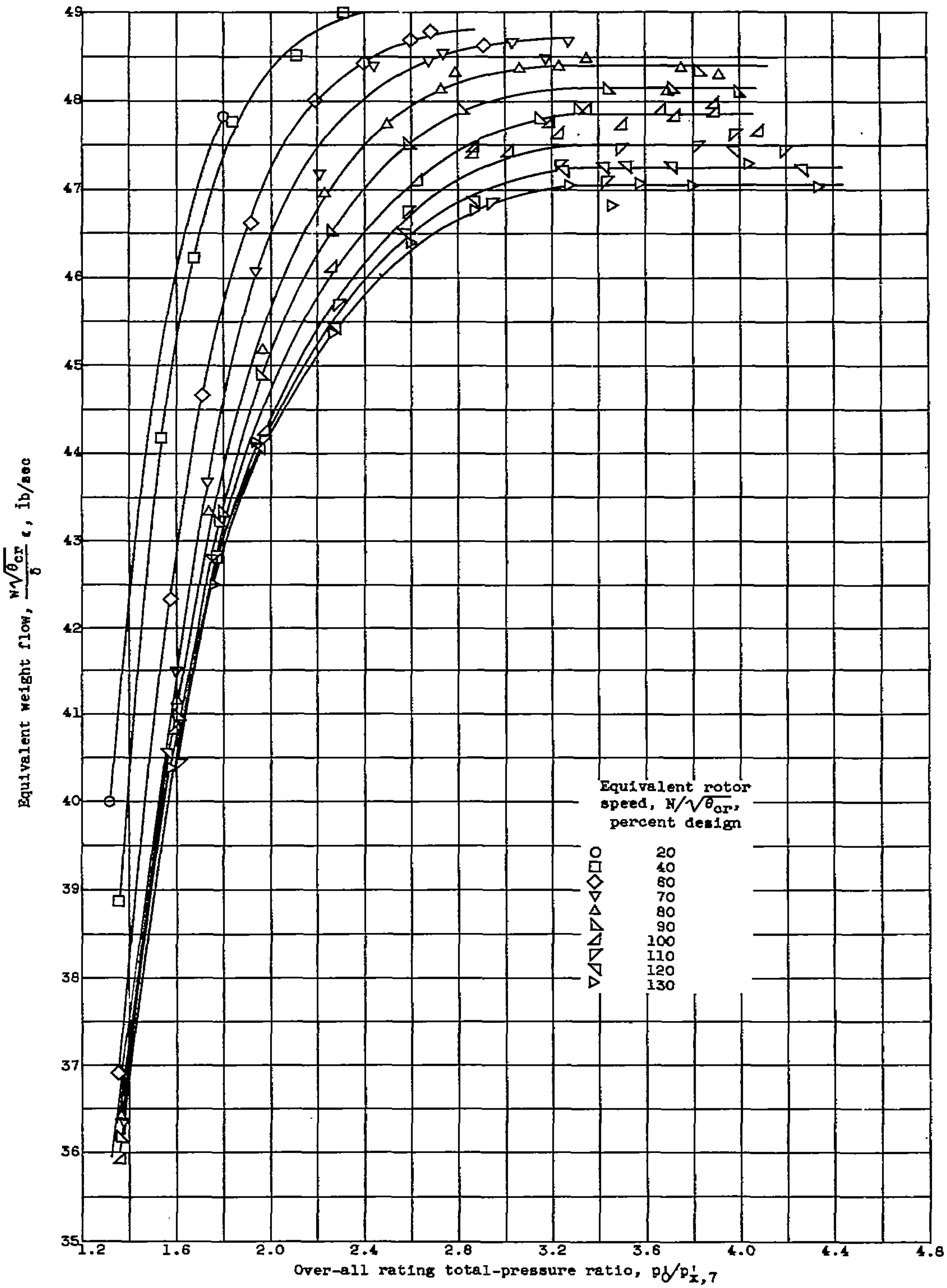


Figure 5. - Variation of equivalent weight flow with over-all rating total-pressure ratio for values of constant equivalent rotor speed.

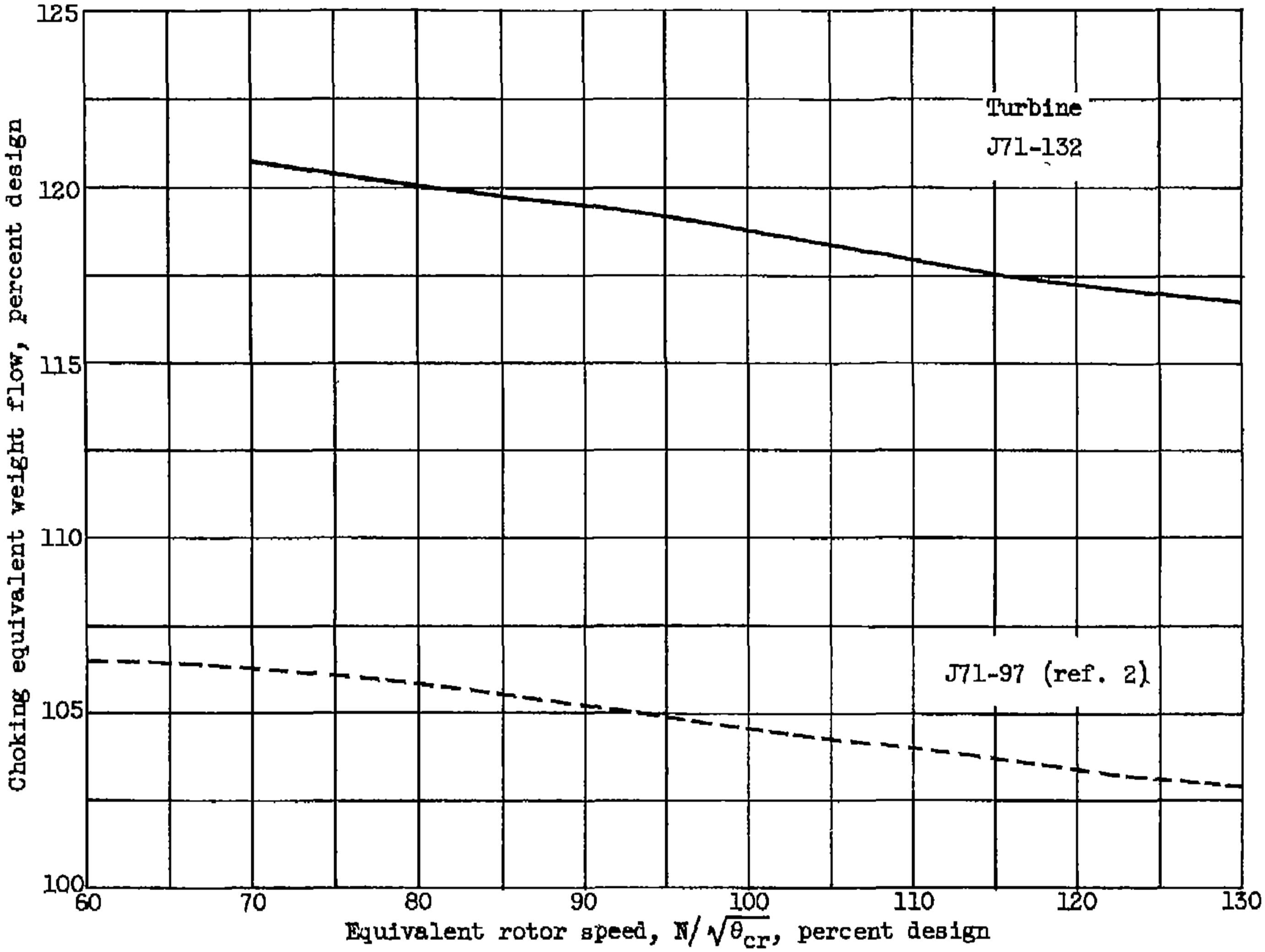
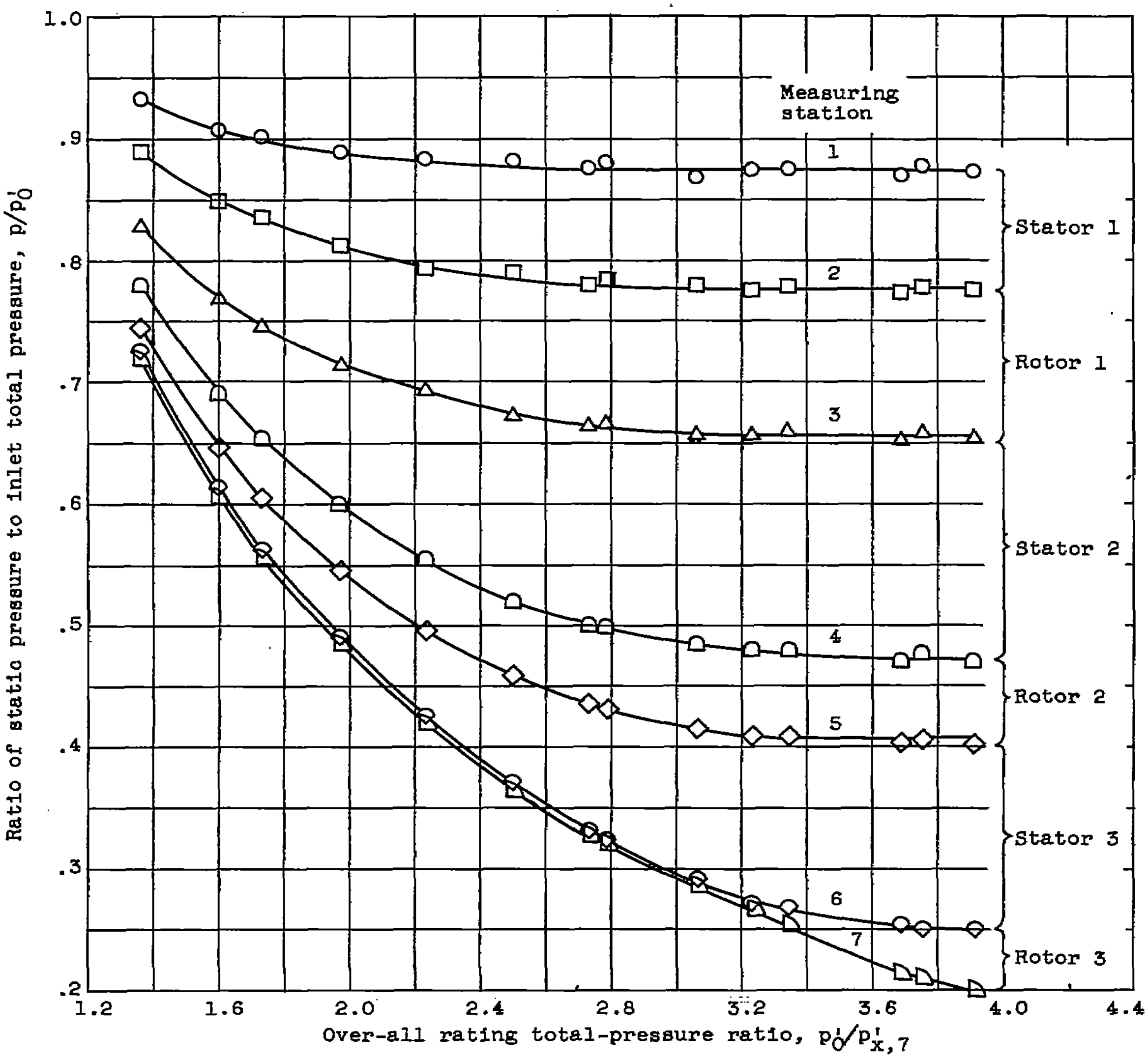
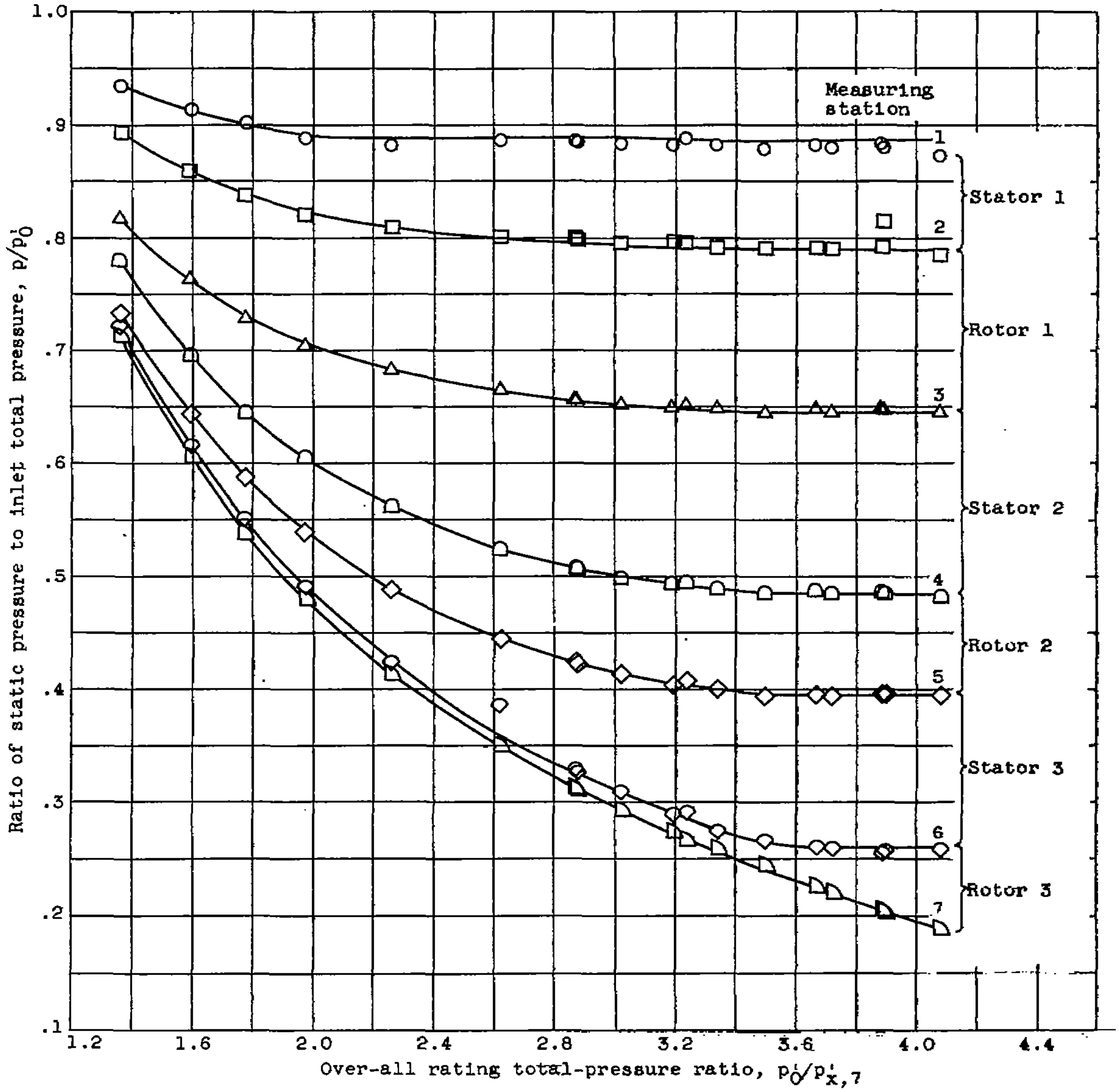


Figure 6. - Variation of equivalent choking weight flow with equivalent rotor speed for two turbine configurations.



(a) Equivalent speed, 80-percent design.

Figure 7. - Variation of static pressure at hub with over-all rating total-pressure ratio at different measuring stations for constant rotor speed.



(b) Equivalent speed, 100-percent design.

Figure 7. - Concluded. Variation of static pressure at hub with over-all rating total-pressure ratio at different measuring stations for constant rotor speed.

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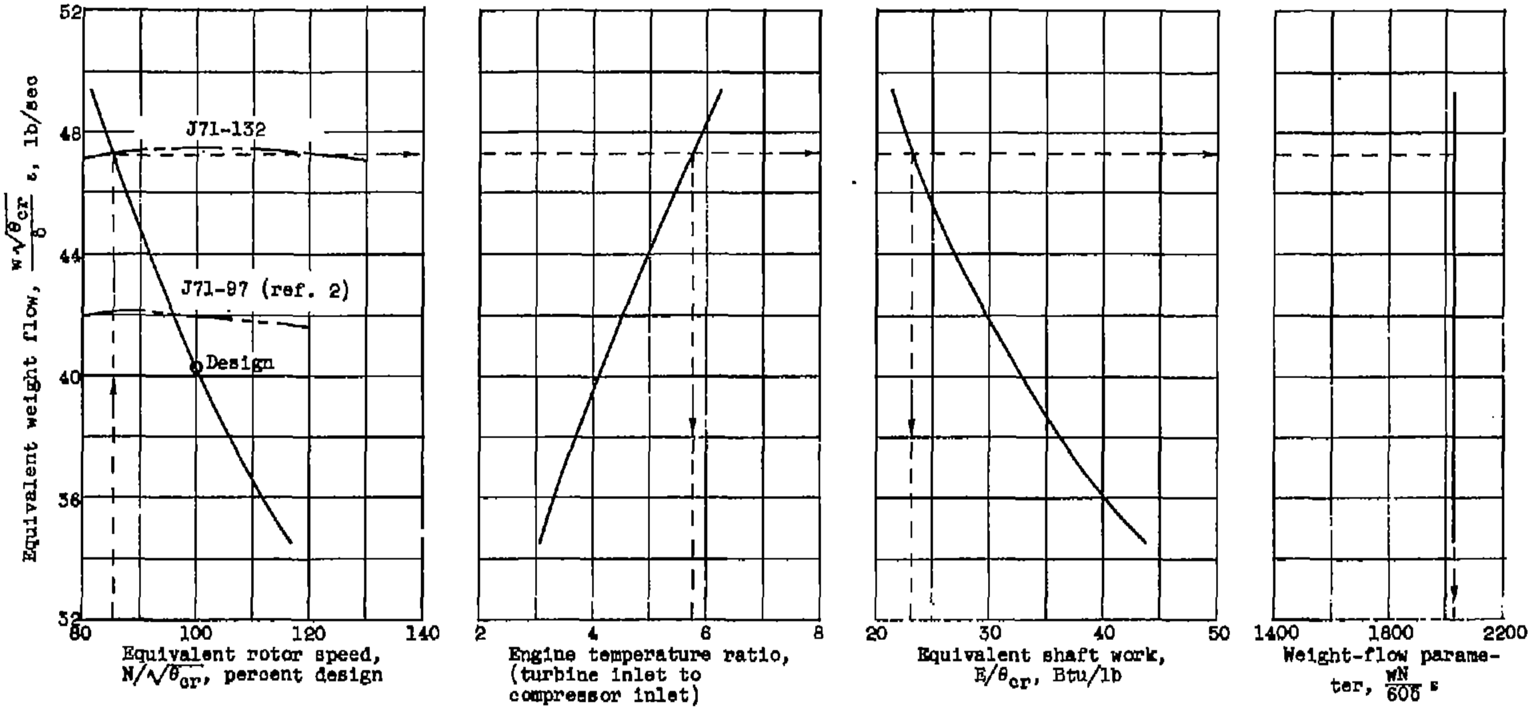

Figure 8. - Curves for determining required turbine operating conditions for maintaining compressor at constant equivalent design conditions.

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